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HIGH FREQUENCY VIBRATIONS ON GEARS

Dietmar Sterns
Head of Engineering, High Speed Gears
RENK Aktiengesellschaft
Augsburg, Germany

Dr. Michael Elbs
Manager of Simulation Engineering
ISMB Dautermann GmbH
Kaufbeuren, Germany

Dietmar Sterns is the Head of the Engineering Department for High Speed Gears of RENK. His responsibilities include mechanical engineering and rotordynamic analysis of High Speed Gear Units. He joined RENK in 1995. Mr. Sterns holds a diploma (Mechanical Engineering, 1995) from the University of Applied Sciences in Augsburg, Germany. He is a member of the Task Forces API 613 (Special Purpose Gear Units) and API 677 (General Purpose Gear Units).

Michael Elbs is the simulation engineering manager of ISMB. His responsibilities include simulations in the fields of structural analysis, vibrations, acoustics and thermal analysis, but also the detailed analysis of measurement data. He joined ISMB in 1998.
Mr. Elbs holds a diploma (Physics, 1994) from the University of Konstanz, Germany and a doctorate degree (natural sciences, 1998) from the University of Hannover, Germany

ABSTRACT

Accelerometers are commonly used on high speed gear units that are built according to API 613. But the use of the signals from the accelerometers is very often subject to discussion. According to the experience of the authors, the questions discussed the most are:

- Should the accelerometer signal be used as acceleration or velocity?
- Should trip points be based on RMS (Root Mean Square) or 0-peak values?
- What frequency range should be analyzed?
- What values should be used for the alarm and trip set points?

There are no general answers to these questions. Especially for acceleration signals, the choices are complex and need to be analyzed case by case. The intention of this paper is to help engineers to better understand acceleration signals from High Speed Gear Units and to give recommendations on how to use and analyze such signals.

Field data recorded on many different production machines indicated that the total acceleration value can vary strongly depending on speed. It is presumed that this is due to resonances excited by the gear mesh frequency (see Figure 1). In some of these resonances, the frequency spectrum of the acceleration signal was dominated by sidebands of the gear mesh frequencies, which could not be explained at first.

This paper presents experimental and analytical investigations on the high frequency vibration behavior (accelerations) measured on gear casings. Detailed investigations were performed on gear units designed for motor-compressor applications. The outcome of these investigations is an analytical approach that explains the reason for the observed resonances and which predicts the speeds at which these resonances occur as well as the frequencies of the side bands that occur in the proximity of the gear mesh frequency during such a resonance. Additionally, this paper will give directions and recommendations on how to use accelerometer signals for condition monitoring of gear units.

INTRODUCTION

Vibrations with gear mesh frequency are a typical signature of any gear unit. On high speed gear units, the gear mesh frequency is usually in the range of 5 to 10 kHz. These vibrations can be measured as acceleration signals. Over the last years, measuring acceleration has become more and more common. But there is still much to learn about the best way to use the vibration acceleration data measured on gear units for monitoring and for machinery diagnosis.

Field data

Several cases were reported from field, where total acceleration amplitudes vary strongly, depending on speed. One example is shown in Figure 1. While the acceleration amplitude is mostly between 4 and 7 g, it increases to much higher values at certain speeds, the highest value being approx. 33 g. Assessments indicated that natural vibration modes of a rotor, which get excited by the gear mesh frequency, could be the cause for the observed acceleration readings. However, a deeper analysis was not possible in this case. More examples of field data are shown in appendix A.

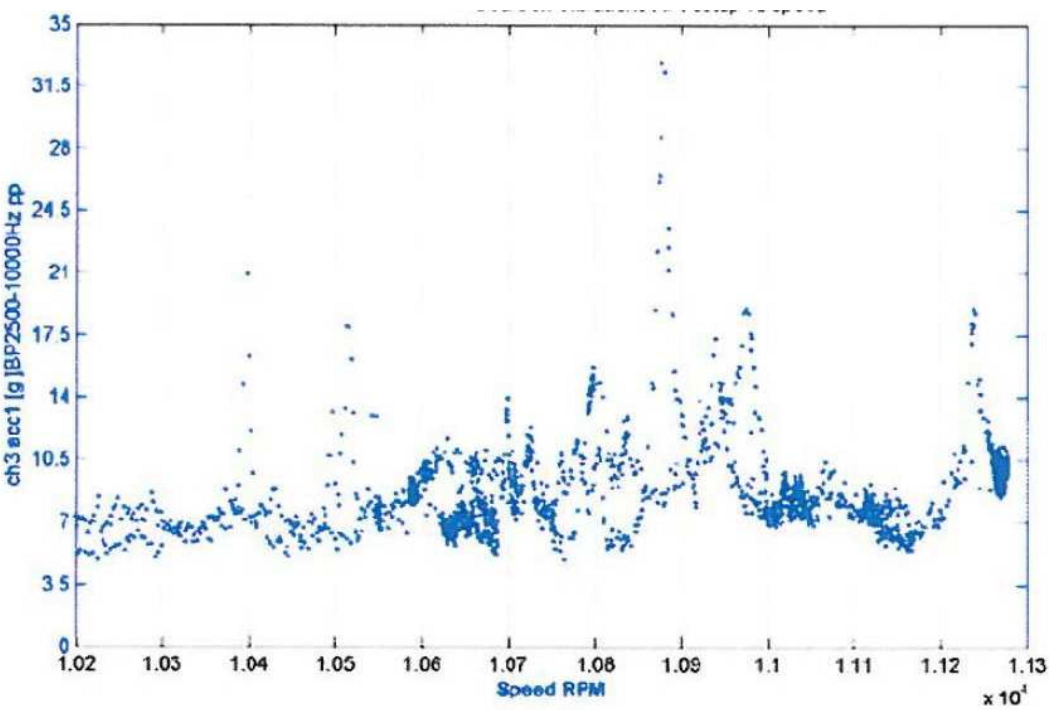

Figure 1. Plot of accelerometer signal vs. compressor speed on a gear unit installed between a motor and a compressor

To learn more about the subject of high frequency vibrations and the resonances measured with accelerometers on gear unit casings, comprehensive investigations were performed. These investigations included detailed vibration measurements on gear units during Factory Acceptance Test (at no load) and during additional tests with load as well as analytical simulations on the effect of these high frequency vibrations on the gear rating and on the durability. The tests were performed on speed increasing gear units, typical for motor-compressor applications. Resonances were detected at very narrow speed bands. In addition to these operating measurements, experimental modal analyses of the gear shafts were performed. By means of detailed FEA models and by considering the Doppler effect related to the relative movement (rotation) of the bull gear (the solid body in resonance) and the exciter (the gear mesh), the speeds at which resonances occur were predicted with high accuracy. In addition, the frequency of the sidebands detected in the frequency spectrum of the accelerometer signal were explained by taking into account the Doppler effect related to the relative movement (rotation) of the bull gear (the solid body in resonance) and the observer (the sensor). Using this FEA model and the data collected during the tests, the additional forces in the gear mesh, which are caused by the observed high frequency vibrations, were calculated and their effect on gear durability was analyzed in accordance with the rating rules of API, AGMA and ISO.

INVESTIGATIONS ON HIGH FREQUENCY VIBRATIONS

Differences between High Frequency Vibration and Low Frequency Vibration

Shaft vibrations typically occur in the low frequency range between 10 Hz and 500 Hz. To measure and evaluate them, engineers commonly look at the displacement amplitude of these vibrations, measured by means of displacement probes. The corresponding acceleration amplitudes are very low. Example: assuming a harmonic vibration at 25 Hz with a displacement amplitude (peak-peak) of 0.4 mils (10 micron), the corresponding acceleration amplitude would be 0.012 g. If the same displacement was observed at a frequency of 200 Hz, the corresponding acceleration amplitude (0-peak) would be 0.8 g.

Gear mesh frequencies on High Speed Gear Units typically occur in the range of 5000 to 10000 Hz. At these frequencies, vibrations have very low displacement amplitudes, but they have high acceleration amplitudes. For example: at a frequency of 5000 Hz, an acceleration amplitude (0-peak) of 10g would correspond to a displacement amplitude (peak-peak) of 0.0078 mils (0.2 micron). At 10,000 Hz, an acceleration amplitude of 10g (0-peak) would correspond to a displacement amplitude of 0.002 mils (0.05 micron). Such displacement amplitudes are negligible; the major effect of such high frequency vibrations is noise.

Using vibration signals for machinery protection and condition monitoring

In the turbomachinery industry, shaft vibrations (displacements) are well understood. The theoretical backgrounds, the rules for protecting machinery by means of alarm limits, and the rules for machinery diagnosis based on detailed analysis of shaft vibration displacement data are all commonly known. Some standards, many books and a vast number of publications are available on this subject.

On the other hand, there is little information available about how to use acceleration signals for machinery protection or condition monitoring. There is no standard and no publication available that would specify accelerometer alarm values for the operation of turbomachines. During the last years, the sensors and equipment for measuring acceleration have developed continuously. The maximum frequency at which acceleration can be measured has increased strongly; 20 kHz can be seen quite often. Therefore, multiples of the Gear Mesh Frequency could be inside the frequency range of the measurement and, consequently, the total value of vibration acceleration measurement would be considerably higher than the value measured in a the frequency range not that large and not including those multiples of Gear Mesh Frequency. An increasing amount of acceleration data is available, while the engineers have no rules at hand to evaluate them. This can lead to long discussions between engineers with different opinions. API Standard 670 recommends the use of accelerometer signals on gear units for condition monitoring but not for machinery protection. The outcome of the tests and analyses described in this paper are supporting this recommendation of API standard 670. More detailed recommendations are given in the conclusions of this paper.

Tests and measurements

A double helical gear unit with a rated power of 3.9 MW was equipped with additional vibration sensors for specific tests. The bull gear has 239 teeth and the pinion has 28 teeth. At the rated speed of 1800 rpm, the gear mesh frequency is 7170 Hz. Figure 2 shows the test arrangement as used in the gear manufacturer’s shop. The driver was a motor with a rated power of 1 MW. The brake was a generator with a rated power of 1 MW.

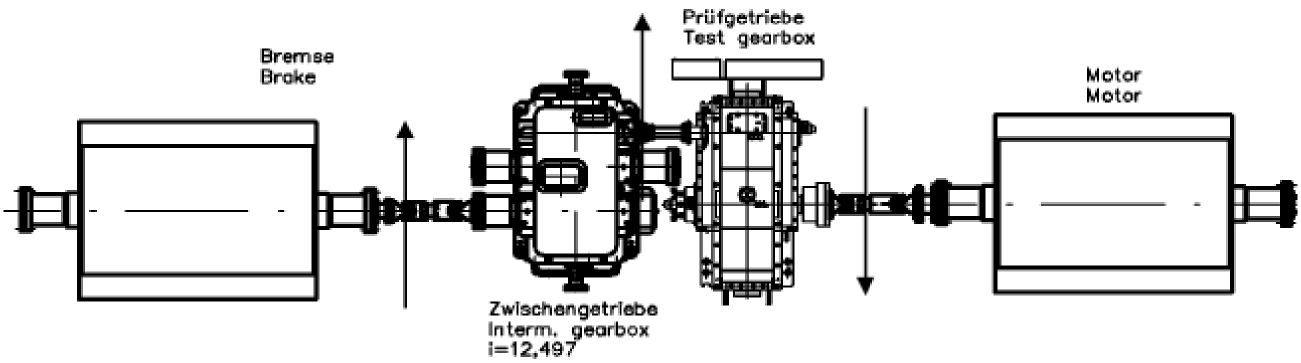

Figure 2. Test arrangement

In addition to the 2 job accelerometers, 4 more accelerometers and one displacement probe were installed (see Figure 3 and Figure 4). The job acceleration sensors were installed in the positions MP1 and MP4. The additional displacement probe was installed in an axial direction facing the side of the bull gear disc close to its outer diameter (see Figure 4).

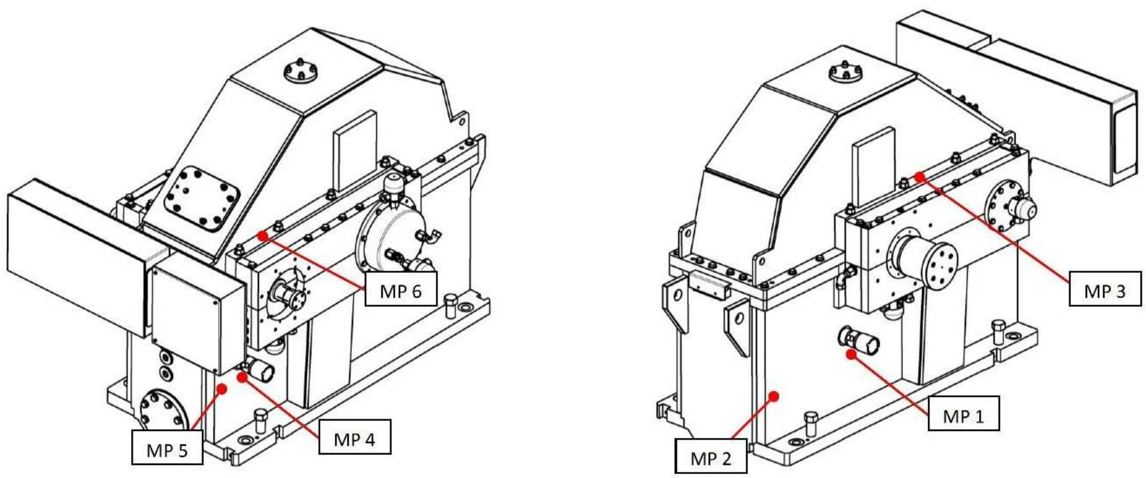

Figure 3. Positions of the acceleration sensors

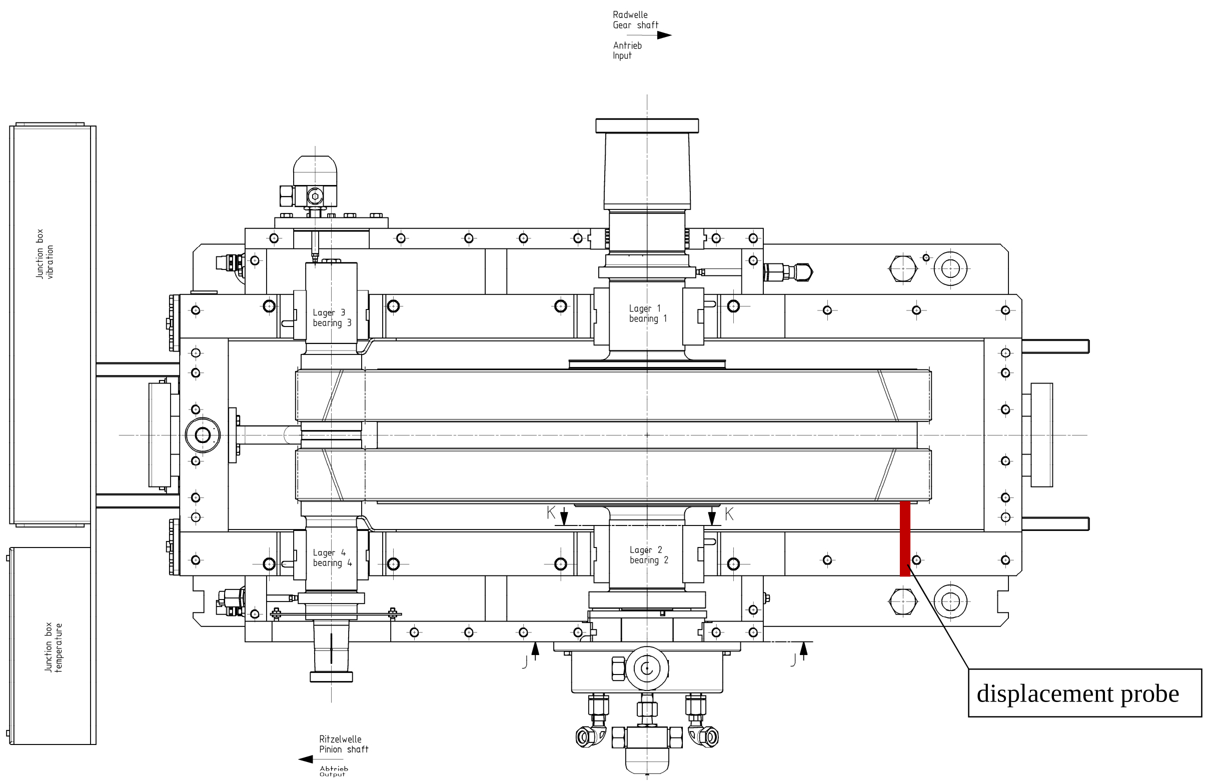

Figure 4. Cross-section of the gear unit used for the tests

The gear unit was loaded with a constant torque corresponding to 25% of the rated torque. The speed was varied from 800 rpm to 1900 rpm; the rate of speed change was 1 rpm/s. At a couple of speeds, the acceleration signal (total value) increased remarkably, indicating some kind of resonance, as shown in Figure 5. A low rate of speed change is necessary in order to detect these resonances. At higher rates of speed change (i.e. faster speed acceleration or deceleration), the accelerometer signal did not display such peaks.

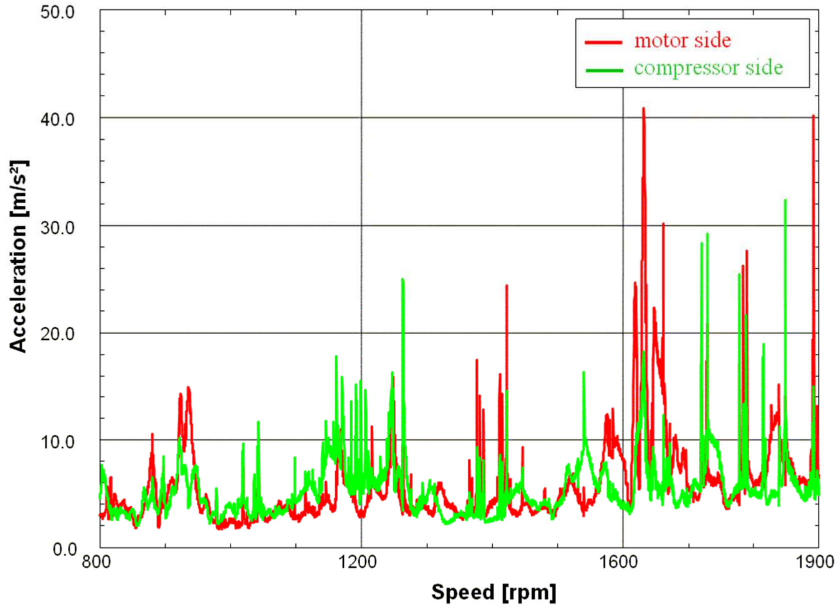

Figure 5. Total value (rms) of acceleration vs. speed, sensors MP1 (motor side) and MP 4 (compressor side)

Some of the peaks occur in a very narrow speed range of less than 1 rpm, indicating very little damping, like for example the peaks at 1783 rpm and 1891 rpm – see Figure 6.

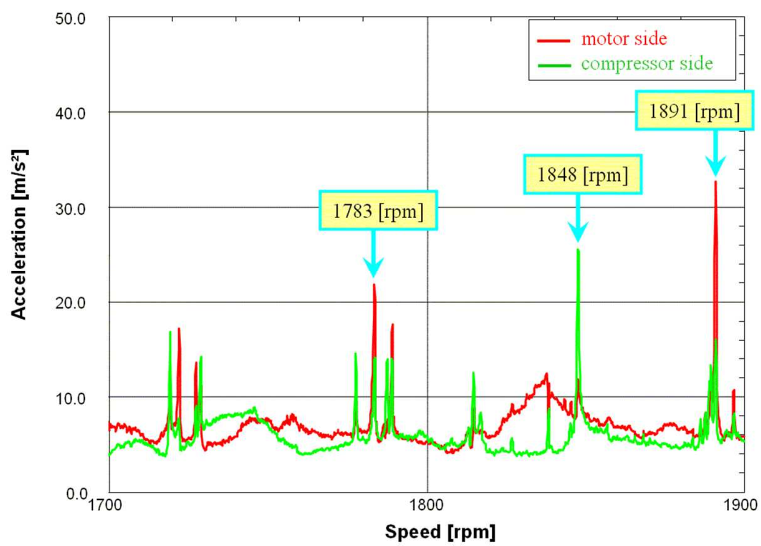

Figure 6. Extract from figure 5: Total value (rms) of acceleration in the speed range 1700 rpm to 1900 rpm, sensors MP1 (motor side) and MP 4 (compressor side)

When the acceleration signal shows a resonance, its frequency spectrum is dominated by the Gear Mesh Frequency (GMF) and its multiples and by certain sidebands of the gear mesh frequency – see Figure 7 with an overview of the frequency spectrum of the acceleration signal.

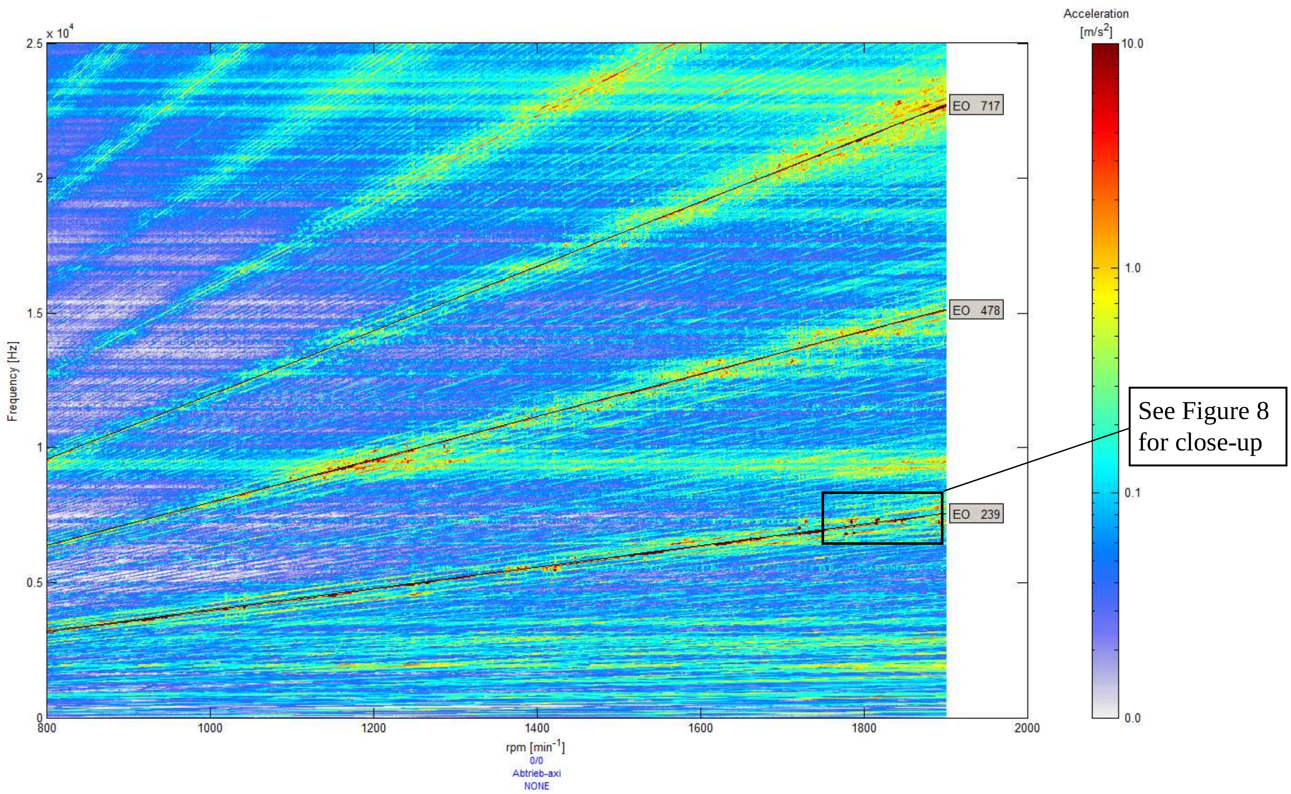

Figure 7. Sonogram of the accelerometer signal, sensor MP1 (motor side)

Note: a sonogram is similar to a waterfall plot, showing the acceleration amplitude versus rotor speed and frequency. In a sonogram, the acceleration amplitude is not indicated by a third dimension, but by color – see the scale on the right side of the diagram.

Surprisingly, in most cases the sidebands were observed only on one side of the Gear Mesh Frequency GMF – either below or above the GMF (see Figure 8). At first, this is unexpected, because classic amplitude or phase modulations would produce two symmetrical sidebands around the GMF. The difference between the sideband frequency and the GMF corresponds to an even multiple of the bull gear rotational frequency. As an example, Figure 8 shows two speeds with resonance (1816 rpm and 1847 rpm) and their main frequency components. At 1816 rpm the dominant frequencies are the Gear Mesh Frequency GMF (order 239, which is identical with the number of teeth on the bull gear z=239) and a sideband at 243X, which corresponds to GMF + 4X. At 1847 rpm the dominant frequencies are the Gear Mesh Frequency GMF (239X) and a sideband at 235X, which corresponds to GMF – 4X. Figure 9 shows the order spectra at 1816 rpm and 1847 rpm respectively. The two resonance speeds and the side bands observed are related to each other. The correlation will be explained in the following section.

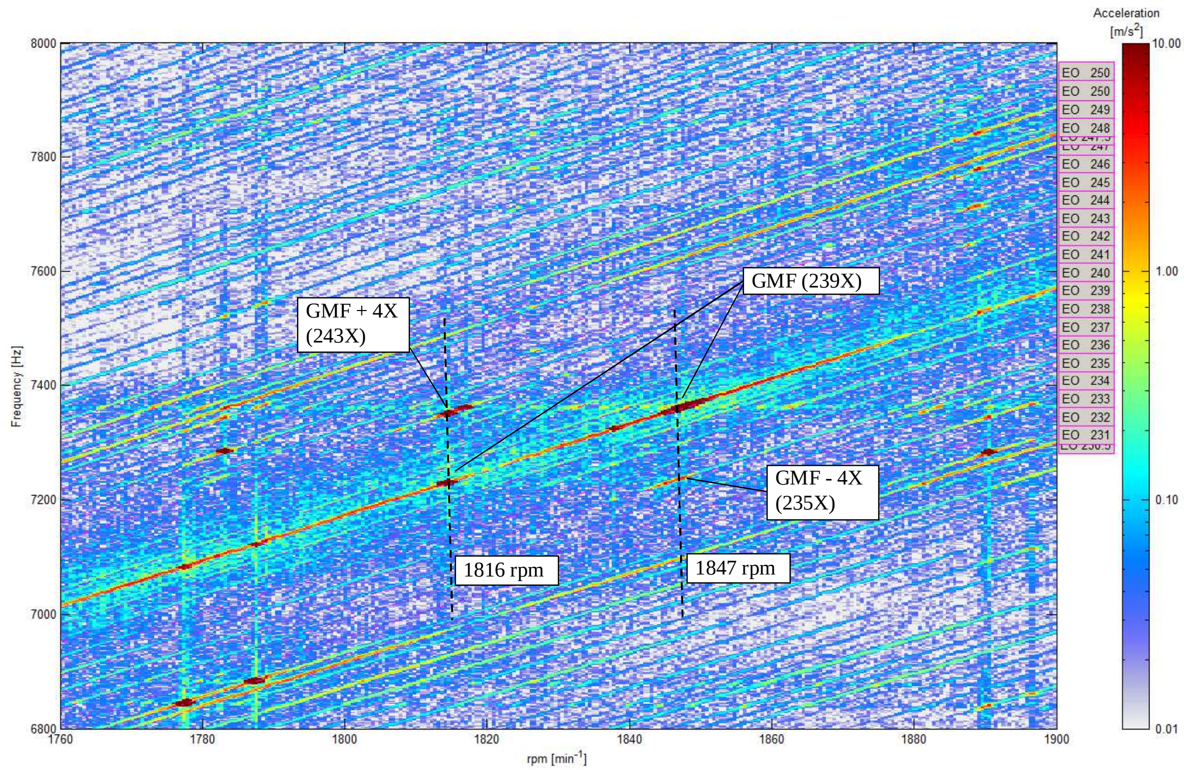

Figure 8. Sonogram of the accelerometer signal (extract from figure 7), example no.1

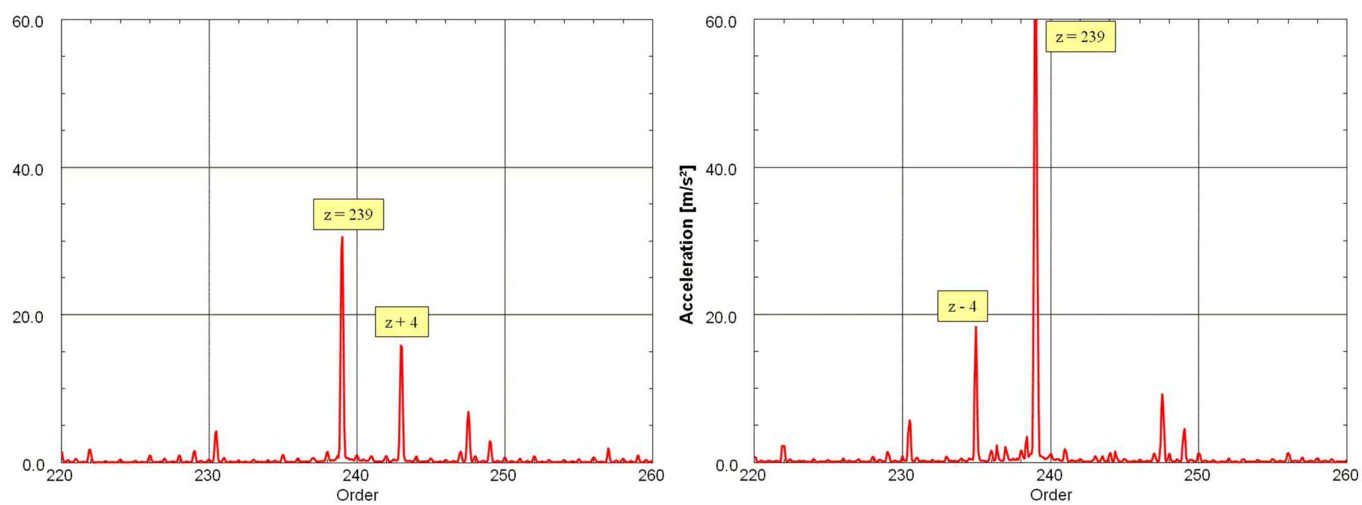

Figure 9. Frequency spectrum of the acceleration signal recorded at 1816 rpm (left) and 1847 rpm (right)

Another example is shown in Figure **10**. There are two further speeds with resonance: one at 1783 rpm and one at 1891 rpm. Three sidebands were observed at each of these two resonance speeds. The order spectra for these two speeds are shown in Figure 11.

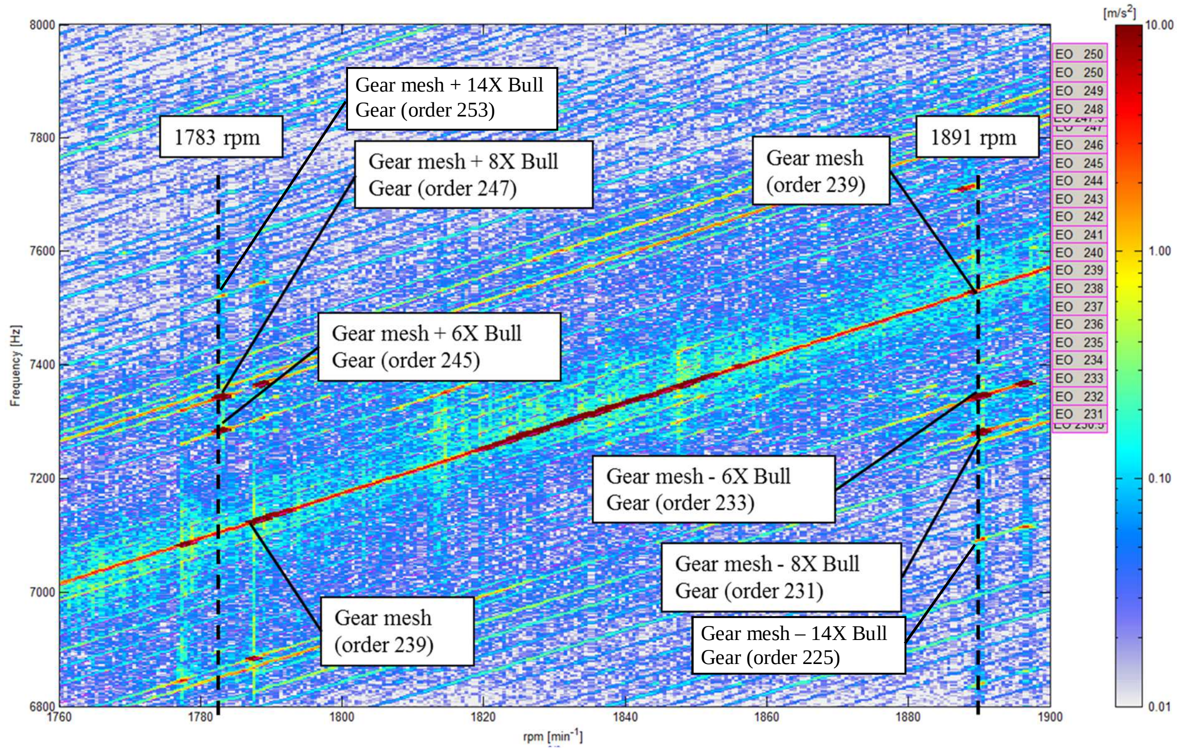

Figure 10. Sonogram of the accelerometer signal (extract from figure 7), example no.2

The plot on the left side of Figure **11** shows the order spectrum of the acceleration signal recorded at 1783 rpm. There is a clear peak at the Gear Mesh Frequency GMF, but there are other peaks at the orders 245 (GMF + 6X bull gear speed), 247 (GMF + 8X bull gear speed) and 254 (GMF + 14X). The order spectrum recorded at 1891 rpm, as shown on the right side of the same figure, contains a peak at the order 239 (GMF), but also other peaks at the orders 233 (GMF – 6X bull gear speed), 231 (GMF – 8X bull gear speed) and 224 (GMF – 14X bull gear speed). The two resonance speeds and the side bands observed are related to each other. The correlation will be explained in the following section.

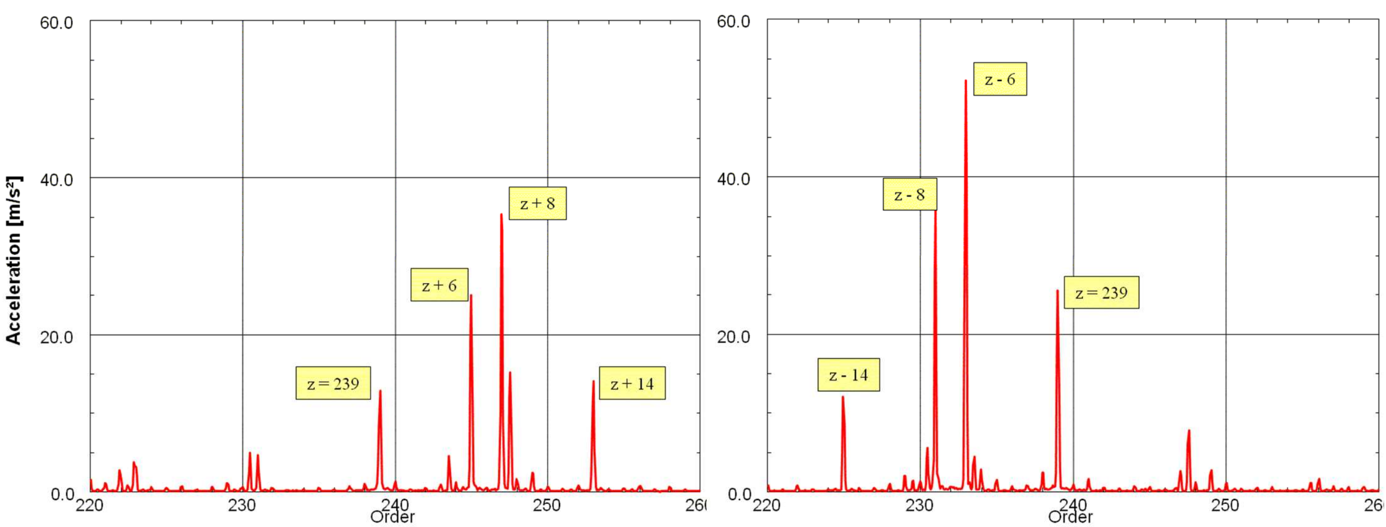

Figure 11. Frequency spectrum of the acceleration signal recorded at 1783 rpm (left) and 1891 rpm (right)

Analytical investigations on resonance speeds and frequency spectrum

A numerical modal analysis was performed with an FEA model of the Low Speed Shaft. To validate this model, an experimental modal analysis (ping test) was performed. The resonance frequencies measured in the experimental modal analysis were in very good correlation with the calculated natural frequencies.
With this validated model, the narrow-band resonances observed during the tests could be correlated with natural frequencies of the bull gear disc regarding their frequency, their damping behavior and their mode shapes. However, the correlation of the measured resonance speeds with the analytically predicted resonance speeds was not nearly as good as for the modal analyses. This difference is easily seen in the plots displaying the amplitude at two times gear mesh frequency versus speed for both measurement and simulation (see Figure 12). Wherever the simulation indicated one resonance speed, the measurements detects two resonance speeds, one lower and one higher than the resonance speed calculated in the simulation. The value of the resonance speed calculated in the simulation was always equal to the mean value of the two resonance speeds detected by measurement.

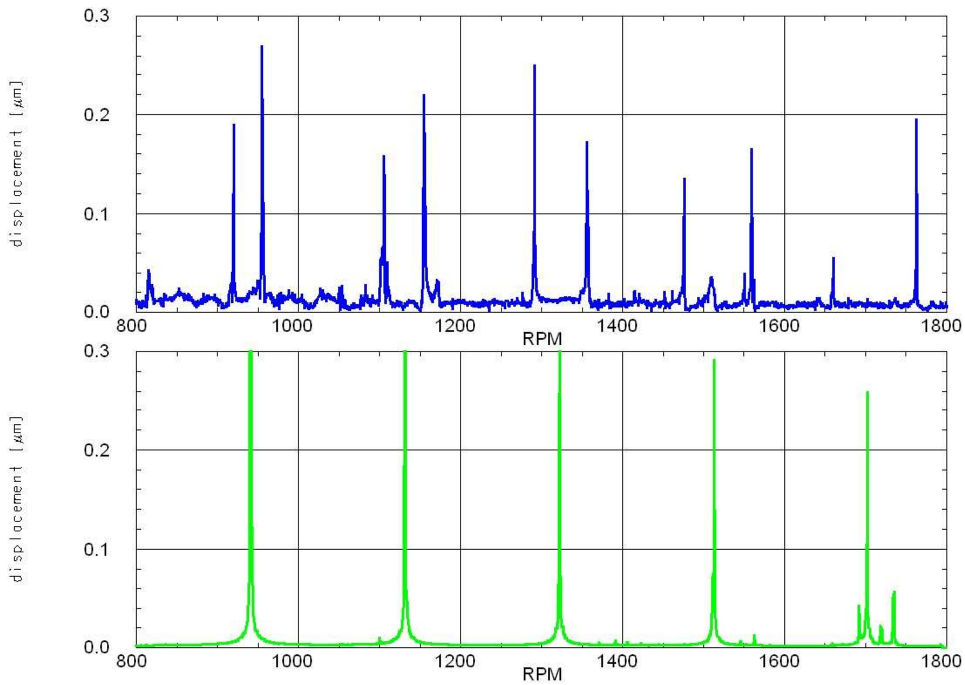

Figure 12. Order section showing the displacement amplitude at the displacement probe on the gear disc at double gear mesh frequency (478X) vs. speed. Comparison between measurement (top) and simulation with the gear wheel stationary (bottom)

The correct value of the resonance speeds – the shaft speeds at which resonance occurred – can be explained and calculated only by taking into account the relative movement between the vibrations of the rotating gear wheel and the locally fixed parametric excitation in the gear mesh. With this approach, the simulation reproduces the measurement results with high accuracy. Two resonance speeds are predicted for each natural frequency of the gear wheel and they match with the measured natural frequencies (see Figure 13).

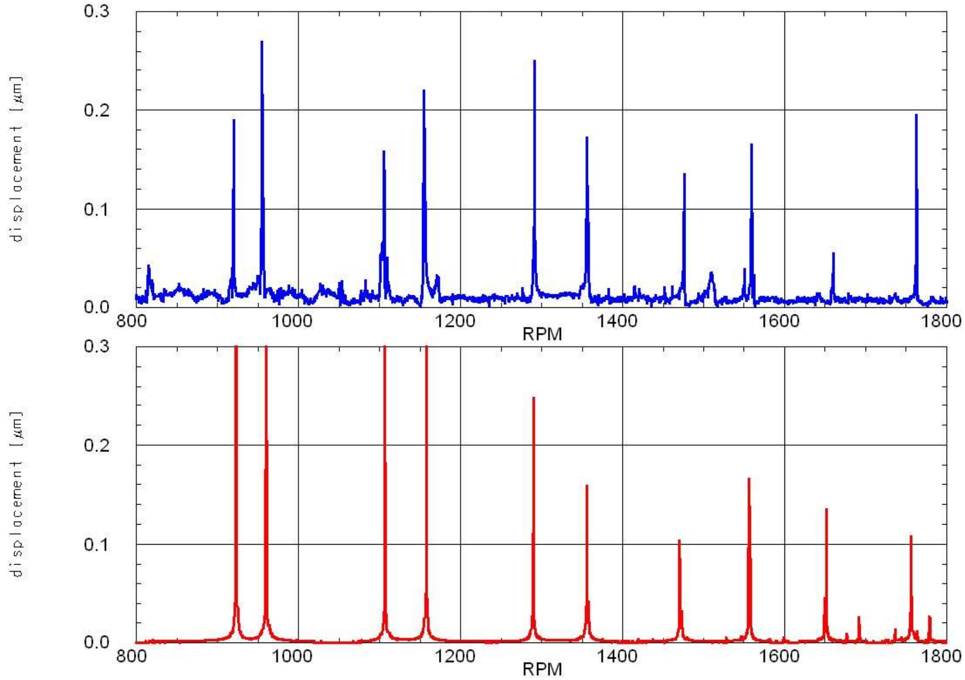

Figure 13. Order section showing the displacement amplitude at the displacement probe on the gear disc at the double gear mesh frequency (478X) vs. speed. Comparison between measurement (above) and simulation with the gear wheel rotating (below)

In terms of mathematics, the aforementioned relative movement can be taken into account via two approaches:

- Apply the equations of the Doppler effect
- Apply the equations of amplitude modulation

Elbs and Sterns (2015) elaborated the mathematical approach via the Doppler Effect in more detail. By considering the effect of amplitude modulation between the resonance frequency of the bull gear as measured or calculated at zero speed and the frequency of the bull gear rotation (which represents the movement of the gear mesh relative to the bull gear), the same results are obtained. The equations to calculate the two speeds of resonance related to a given natural vibration mode of the bull gear disc are:

$$n_1 = \frac{f_0 * 60}{z + k} \qquad \text{Equation 1}$$

$$n_2 = \frac{f_0 * 60}{z - k} \qquad \text{Equation 2}$$

where:

n_1, n_2	speeds at which resonance occurs, rpm
f_0	natural frequency of the natural vibration mode of interest, Hz
k	order number of the natural vibration mode related to the gear mesh (equal to the number of sinus waves on the outer diameter of the gear wheel)
z	number of teeth on the gear

The order number k depends on the mode shape of the mode of interest, especially at the position of the gear mesh (close to the outer diameter of the bull gear disc). As an example, for the mode shape shown in Figure 14, the order number k is equal to 7, as the mode shape forms 7 sinus waves (deflections) along the circumference of the gear wheel.

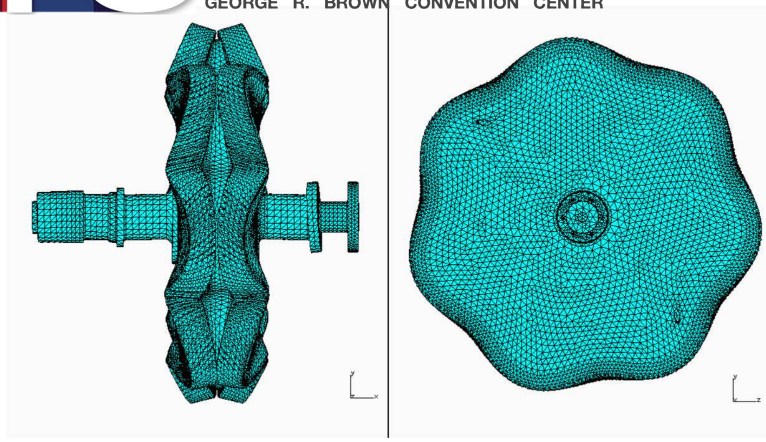

Figure 14. Example of a mode shape with order number k=7 at the outer diameter of the gear wheel and with order number m=1 at the bearings

Furthermore, the relative movement between the vibrations of the rotating gear wheel and the locally fixed accelerometer needs to be considered in order to explain and correctly calculate the frequencies observed in the accelerometer signal. This relative movement can be taken into account via the same two approaches, Doppler Effect or amplitude modulation, as elaborated by Elbs and Sterns (2015). The equations needed to calculate the frequencies are:

for the resonance at the speed n_1:

$$f_{1a} = \frac{n_1}{60} * (z + k + m)$$ Equation 3

$$f_{1b} = \frac{n_1}{60} * (z + k - m)$$ Equation 4

for the resonance at the speed n_2:

$$f_{2a} = \frac{n_2}{60} * (z - k + m)$$ Equation 5

$$f_{2b} = \frac{n_2}{60} * (z - k - m)$$ Equation 6

where:

$f_{1a,b}$	sideband frequencies at the resonance speed n_1
$f_{2a,b}$	sideband frequencies at the resonance speed n_2
n_1, n_2	speeds at which resonance occurs, rpm
z	number of teeth on the gear
k	order number of the resonance mode related to the gear mesh (equal to the number of sinus waves on the outer diameter of the gear wheel)
m	order number of the resonance mode related to the bearings

The order number m relates to the mode shape of the mode of interest, especially to the deflections of this mode shape at the position of the bearings, as this is the place where the vibration is transmitted from the shaft to the casing. In many cases, the order number m equals the order number k. In those cases, the equations for the frequencies related to n1 and n2 are simplified to:

$$f_{1a} = \frac{n_1}{60} * z = GMF_1$$ Equation 7

$$f_{1b} = \frac{n_1}{60} * (z + 2 * k)$$ Equation 8

$$f_{2a} = \frac{n_2}{60} * z = GMF_2$$ Equation 9

$$f_{2b} = \frac{n_2}{60} * (z - 2 * k)$$ Equation 10

The first example of resonance speeds and their frequency spectrum as shown in Figure 8 are related to a natural vibration mode of the gear wheel with the natural frequency f_0=7295 Hz and with the order numbers k=m=2. By using equations 1 and 2, the resonance speeds can be calculated: n_1=1816 rpm and n_2=1847 rpm. The characteristic frequencies can be calculated with the equations 7 thru 10; for n_1=1816 rpm they are equal to 7234 Hz (GMF_1) and 7355 Hz (GMF_1 + 4X), and for n_2=1847 rpm they are equal to 7357 Hz (GMF_2) and 7234 Hz (GMF_2 – 4X).

The resonance speeds and their frequency spectra as shown in Figure 8 and Figure 11 are related to a natural vibration mode of the gear wheel with the natural frequency f_0=7311 Hz and with the order numbers k=7 and m=1. By using equations 1 and 2, the resonance speeds can be calculated: n_1=1783 rpm and n_2=1891 rpm. The characteristic frequencies can be calculated with the equations 3 thru 6; for n_1=1783 rpm they are equal to 7281 Hz (GMF_1 + 6X) and 7340 Hz (GMF_1 + 8X), and for n_2=1891 rpm they are equal to 7343 Hz (GMF_2 – 6X) and 7280 Hz (GMF_2 – 8X).

Analytical investigations of the gear loads caused by the vibrations

To evaluate the dynamic loads in the gear mesh, an FEA model of the complete gear set was created. For this purpose, the model included both the low speed and high speed shafts and the adjoining couplings (see Figure 15). The periodic variation of gear mesh stiffness was considered according to calculation method A described in ISO 6336. This periodical variation of the gear mesh stiffness, which is typical for every gear unit, causes a corresponding periodic (harmonic) variation of the gear mesh forces, synchronous with the gear mesh frequency. This periodic variation of the gear mesh forces can be expressed by a superposition of a constant force (corresponding to the torque that is transmitted through the gear unit) and a dynamic force synchronous with the gear mesh frequency. The dynamic part of the gear mesh forces is the excitation mechanism for the resonance of the bull gear. For gear rating, the possibility of getting dynamic forces superposed on the constant forces is taken into account by means of the Dynamic Factor K_V, which is equal to the ratio between the dynamic part of the gear mesh forces and the constant, average gear mesh force.

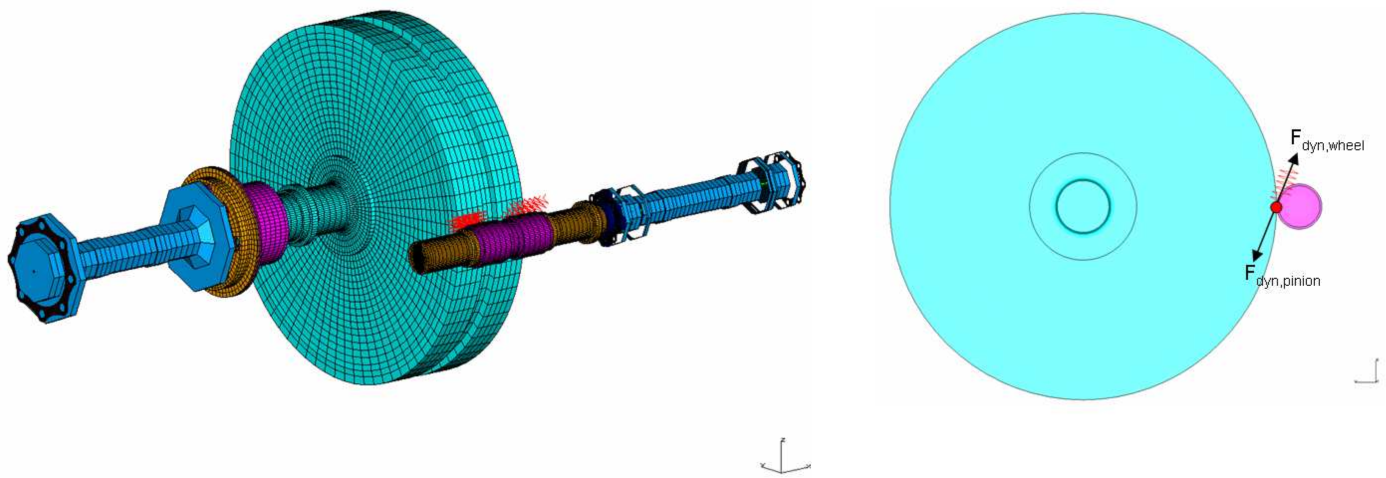

Figure 15. FEA model for calculating the Dynamic Factor Kv

Assuming linear conditions, the dynamic stiffness variation at the gear mesh frequency was transformed to additional dynamic excitation forces acting within the gear mesh. A response analysis due to excitation by these excitation forces was performed. The excitation forces act along the gear mesh direction considering the pressure angle and the helix angle – see Figure **15**. The magnitude of the excitation forces was tuned to match the vibration displacements, which were measured with a displacement probe on the lateral face of the gear disk (see Figure 4). The resulting dynamic forces within the gear mesh were analyzed and compared to the average gear mesh forces. That way, the actual Dynamic Factor Kv was calculated – see Figure 16.

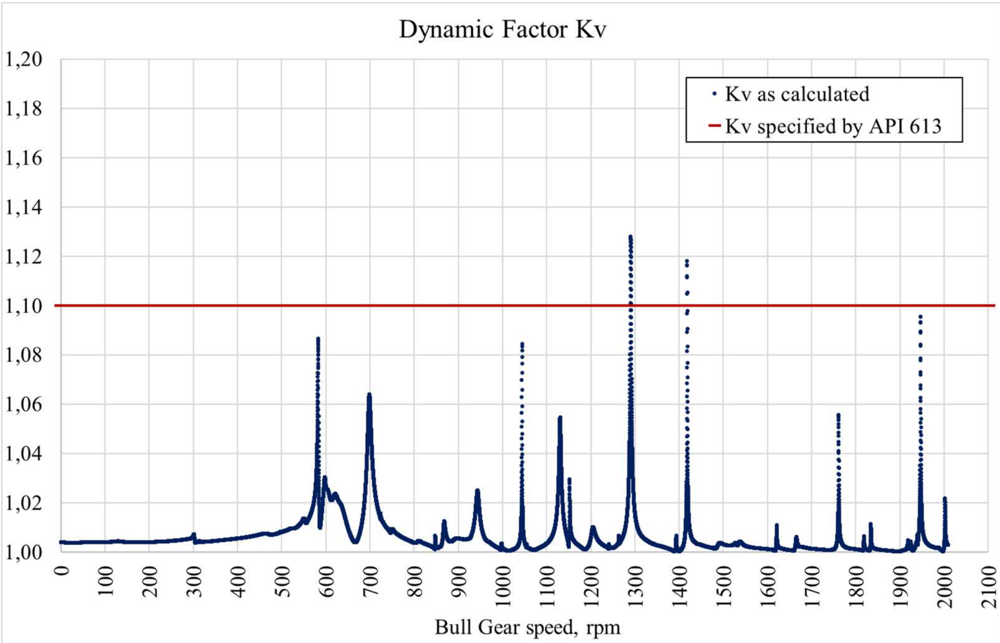

Figure 16. Dynamic Factor Kv versus bull gear speed

API Standard 613 specifies a value of 1.1 for the Dynamic Factor Kv to be used for the gear rating. That value covers the dynamic forces as calculated over the large majority of the speed range. Only at two resonance speeds, the actual Dynamic Factor is higher than the value of 1.1 specified in API 613 to be used for the rating. Using the values of Kv as calculated and the allowable stresses specified in API 613, an actual Service Factor value was calculated versus bull gear speed – see Figure 17. At the two resonance speeds mentioned before, the actual Service Factor is slightly lower than the value of 1.6 specified in API 613. The lowest value calculated is 1.56. That is still enough margin to allow operation of the gear unit without any restrictions. As a summary, the results of those analyzes reveal that the dynamic loads are low and well within the margins given by the rating methods specified in API, AGMA and/or ISO standards. Those rating methods were analyzed by Rinaldo (2017) in comparison to each other. His results demonstrate that API 613 specifies the most conservative gear rating method and allows only 50% to 70% of the load that would be allowable according to AGMA 6011. Thus, the effect of these resonances on the durability of the gears is minor and negligible. The gear rating is not affected, since these resonances are accounted for by the Dynamic Factor Kv.

Vibration data recorded on gear units in the field support these conclusions. As an example, Figure A-2 in the appendix A shows the total peak value of the acceleration signal as recorded on a gear unit installed in a variable speed motor-compressor application, rated for a power of 18 MW. The gear unit operated successfully for more than 25 years without any failure. Despite high acceleration amplitudes that were measured at certain speeds (for example more than 30 g at approx. 10200 rpm), this gear unit had no damage and the gears did not show any sign of wear or crack formation. The plot of amplitudes vs. speed can be considered the baseline of this gear unit.

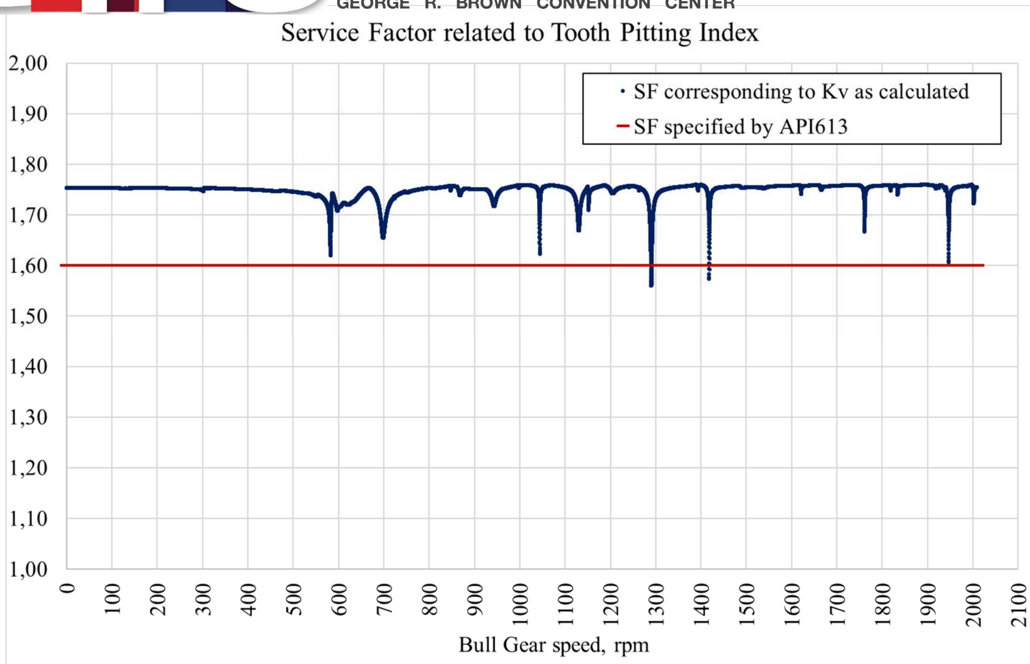

Figure 17. Service Factor deducted from Kv as calculated

CONCLUSIONS

This paper summarizes the theoretical background needed to understand and/or predict high frequency vibrations with gear mesh frequency and the observed sidebands. Additionally, it demonstrates that such vibrations cause only low dynamic stresses on gears, which are within the margins given by the gear rating standards of API, AGMA and ISO and thus will not affect the gear rating. Thus, it enables users to better understand and assess the signature of acceleration signals measured on gear units, especially on variable speed units, which is a premise for doing machinery diagnosis and/or condition monitoring.

The acceleration signal measured with accelerometer sensors on gear units can vary strongly with speed and this can be an indication of a resonance caused by a natural vibration mode of one of the rotors with its natural frequency close to the Gear Mesh Frequency. In general, this is the normal behavior of the gear unit and not an indication of potential damage.

The frequency spectrum of the acceleration signal measured while one of the rotors is in a high frequency resonance will show sidebands next to the Gear Mesh Frequency. The difference between GMF and the frequency of a sideband is equal to a multiple of the speed of the rotor that is in resonance.

Such resonances are possible on any rotor. While this paper focuses on the resonance of the bull gear shaft, the pinion shaft can show resonances too. In that case, the difference between the gear mesh frequency and the frequencies of sidebands will equal a multiple of the pinion shaft speed.

Of course, resonances of the gear unit casing are possible too. In that case, the frequency spectrum of the acceleration signal will have the dominant peak at the gear mesh frequency and will not show high amplitudes at sidebands.

Recommendations

When casing vibrations will be measured on gear units, first decide between measuring velocity or measuring acceleration. Velocity should be measured in the frequency range of 10 Hz to 1000 Hz and displayed as an RMS value (refer to ISO 10816). Acceleration should be measured in the frequency range up to 10000 Hz and displayed as a 0-peak value.

By measuring velocity, low frequency vibrations are monitored – similar to the shaft vibrations. This allows detecting issues like unbalance, misalignment, oil whirl, etc. Velocity signals can be used for machinery protection, condition monitoring and/or machinery diagnosis.

Measuring acceleration is recommended to monitor high frequency vibrations like vibrations close to the gear mesh frequency. Acceleration measurement can help in condition monitoring and machinery diagnosis, but is not recommended for machinery protection.

To be able to use the acceleration signal for condition monitoring, it is necessary to:

- Record the vibration baseline data during a loaded test (if such test is performed) and during commissioning. To record the baseline completely, vary the speed and power over the whole specified range or at least over the expected operating range.
- Store the baseline data. This is the vibration signature of the gear unit in new, undamaged condition and is going to be used as a reference for any vibration data that will be recorded later, during operation.
- Compare any acceleration values measured during operation with the baseline value recorded at the <u>same</u> operating conditions (speed, power, lube oil pressure, lube oil temperature).
- If the total acceleration value recorded during operation shows a strong increase related to the baseline data recorded at the <u>same</u> operating conditions, then the vibration data should be investigated in detail. Ask the OEM and/or a specialist on machinery diagnosis for support.

In general, do not use the total value of the acceleration signal for machinery protection, i.e. for triggering a trip signal. However, the total value of the acceleration signal can be used for machinery diagnosis. It may be used to trigger an alarm, but the alarm value should depend on operating parameters such as speed and power. This variable alarm value should be set in accordance with the baseline values.

The value of the acceleration signal in terms of condition monitoring can be increased by applying frequency filters on the acceleration signal, so that only the frequency band of interest is monitored. For example, a frequency band around the gear mesh frequency may be monitored. For fixed speed applications, this can be achieved quite easily, as the frequency filters can be set to constant values. For speed variable drives, the limit values of the frequency filters need to be proportional to the speed. More complex evaluation methods of the acceleration signal could increase further its value for condition monitoring. Such methods exist and are already in application for roller bearings. For gears they need to be developed respectively to be integrated into monitoring systems.

NOMENCLATURE

f_0	natural frequency of the natural vibration mode of interest, Hz
$f_{1a,b}$	sideband frequencies at the resonance speed n_1
$f_{2a,b}$	sideband frequencies at the resonance speed n_2
GMF	Gear Mesh Frequency, Hz
k	order number of the natural vibration mode of interest (equal to the number of sinus waves related to this mode)
m	order number of the natural vibration mode related to the bearings
n_1, n_2	speeds at which resonance occurs, rpm
z	number of teeth on the gear

FIGURES

APPENDIX A

Example 1:

Gear unit type:	TA50X
Rated power:	1560 kW
Rated input speed:	1500 rpm
Rated output speed:	13957 rpm
Teeth no.:	23/214
Ratio:	9.304
Date of delivery:	July 2004

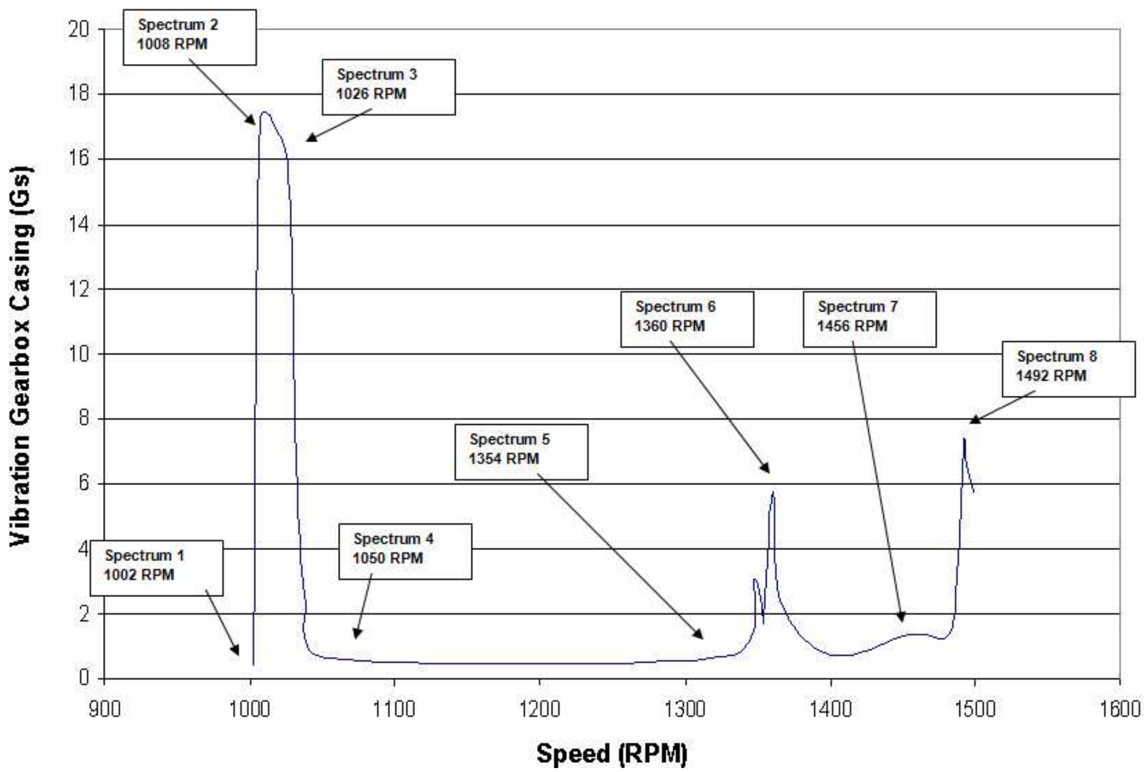

Figure A-1: Casing vibration acceleration in g (rms) vs. input speed

The gear unit in example 1 was operated in the speed range 1000 rpm to 1050 rpm for a couple of weeks after commissioning, as the plant was not running with full capacity yet. The gear mesh was inspected and was found to be in very good condition, no signs of any damage.

Example 2:

Gear unit type:	CD-50
Rated power:	18250 kW
Rated input speed:	3600 rpm
Rated output speed:	10894 rpm
Ratio	3.026
Teeth no.:	38/115
Date of delivery:	1981

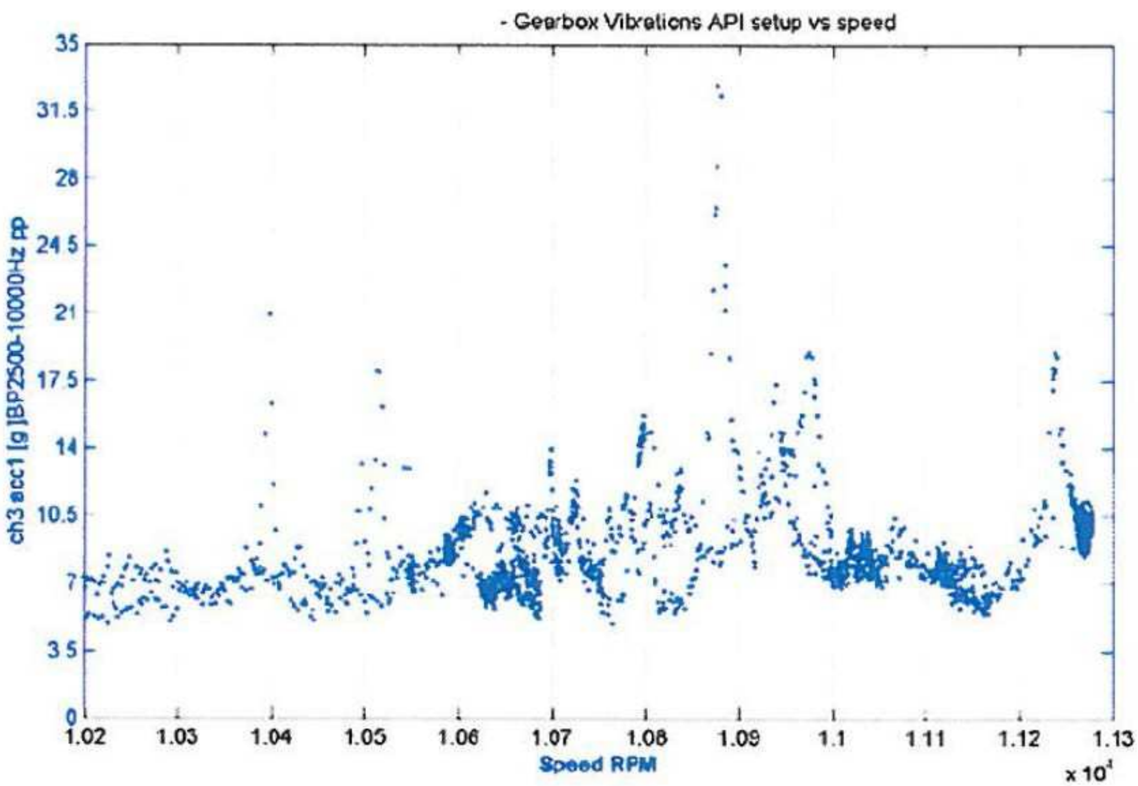

Figure A-2: Casing vibration acceleration in g (peak) vs. output speed (10200 to 11300 rpm), recorded during operation. At around 10880 rpm, a peak value of approx. 33 g (peak) was measured

The gear unit in example 2 was operated for 28 years without any speed range restrictions, while no casing vibration measurement was installed. During measurements in 2009, casing vibration peaks of 80 g (peak) were detected at 10200 rpm. As a consequence, a speed window was programmed in the control software around this speed of 10200 rpm. The gear mesh was inspected and was found to be in very good condition, no signs of any damage.

Example 3:

Gear unit type:	CD-50
Rated power:	21300 kW
Rated input speed:	3600 rpm
Rated output speed:	10718 rpm
Ratio	2.977
Teeth no.:	44/131
Date of delivery:	2008

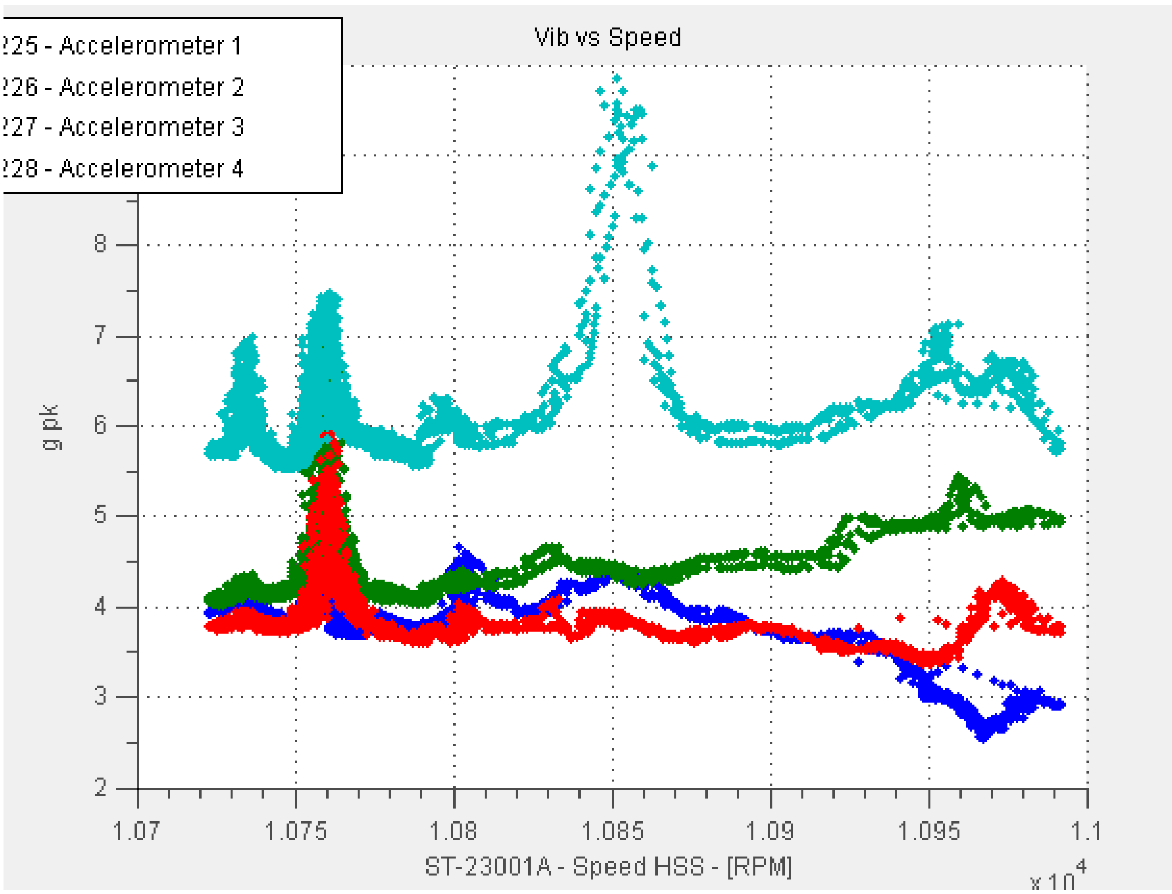

Figure A-3: Casing vibration acceleration in g (peak) vs. output speed 10700 to 11000 rpm. At around 10850 rpm, a peak value of approx. 10 g (peak) was measured.

The gear unit in example 3 is being operated without any speed range restrictions. The gear mesh was inspected and was found to be in very good condition, no signs of any damage.

Example 4

Gear unit type:	TA32X
Rated power:	11000 kW
Rated input speed:	7900 rpm
Rated output speed:	10926 rpm
Ratio	1.383
Teeth no.:	65/47
Date of delivery:	1987

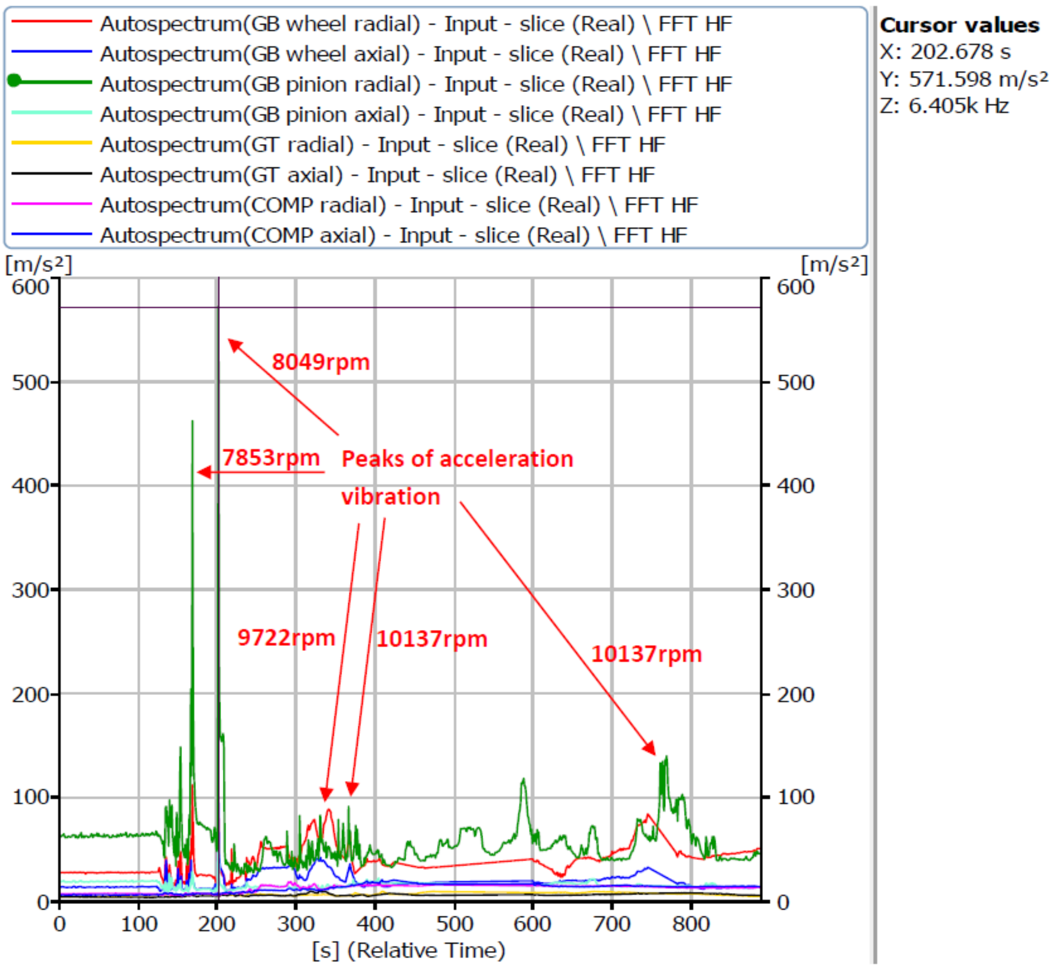

Figure A-4: Casing vibration acceleration in g (rms) vs. time, recorded during start-up and during operation at different speeds. At 8049 rpm, a peak value of approx. 58 g (rms) was measured

The gear unit in example 4 is installed in a variable speed compressor train. It was initially operated without any speed range restrictions, with no gear casing vibration measurement. In 2016, accelerometers were installed and acceleration data were recorded as shown in figure A-4. The gear mesh was inspected and was found to be in very good condition, no signs of any damage.

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